DECLARATION

I, Takeshi MIZUNUMA, a citizen of Japan, 1-20-16, Higashi-ohnuma Sagamihara-shi, Kanagawa Japan, do hereby sincerely declare:

(1) That I am well acquainted with the Japanese Language and English Language, and

(2) That the attached is a full, true and faithful translation into the English language made by me of the certification of the Japanese Patent Application No. 2002-213785 filed on July 23, 2002.

This January 27th, 2005 at Kanagawa, Japan

Takeshi Mizunuma

Takeshi MIZUNUMA

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[Title of the Invention] Rolling Bearing for belt-type continuously variable transmission

[Claims] 1

[Inventor]

Address: c/o NSK LTD., 1-5-50, Kugenumashinmei, Fujisawa-shi, Kanagawa-ken, Japan

Name: Hiromichi TAKEMURA

[Inventor]

Address: c/o NSK LTD., 1-5-50, Kugenumashinmei, Fujisawa-shi, Kanagawa-ken, Japan

Name: Yoshiaki SAKAJIRI

[Applicant]

ID No.: 000004204

Name: NSK Ltd.

[Attorney]

ID No.: 100087457

Name: Takeo KOYAMA

[Nominated Attorney]

ID No.: 100120190

Name: Shun NAKAI

[Nominated Attorney]

ID No.: 100056833

Name: Kinzo KOYAMA

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[Name of Document] Specification (2002-213785)

[Title of the Invention] Rolling Bearing for Belt-type continuously variable transmission

[Scope of The Patent Claim]

[Claim 1] A rolling bearing for belt-type continuously variable transmissions comprising: an outer ring having an outer ring raceway on an inner peripheral surface; an inner ring having an inner ring raceway on an outer peripheral surface; and a plurality of rolling elements rotatably provided between the outer ring raceway and the inner ring raceway, characterized in that, when a minimum thickness of a part where the outer ring raceway is provided on a middle portion in the axial direction of the outer ring is h, and a diameter of each rolling element is Da, the relationship $0.4Da \leqq h \leqq 0.8Da$ is satisfied.

[Detailed Description of the Invention]

[0001]

[Technical Field to Which the Invention Belongs]

The present invention is related to the improvement of rolling bearings for supporting a belt-type continuously variable transmission for automobiles. More specifically speaking, even if as a low-viscosity fluid is used as CVT fluid (including ATF compatible oil), and incorporated into transmission cases of low rigidity, sufficient durability can be maintained.

[0002]

[Prior Art]

As disclosed in, for example, Japanese Examined Utility Model Publication No. H08-30526 and the like, a variety of belt-type continuously variable transmissions has been heretofore designed as speed changing units in automatic transmissions for automobiles, and some of these have been employed in practice. FIG. 2 shows the basic structure of such a belt-type continuously variable transmission in simplified form. This belt-type continuously variable transmission has an input rotating shaft 1 and an output rotating shaft 2 which are mutually arranged in parallel. These rotating shafts 1 and 2 are rotatably supported within a transmission case (not shown in drawings), which is a fixed part, by respective pairs of rolling bearings 3, 3.

[0003]

As shown in detail in FIG. 3, each rolling bearing 3 has a concentric outer ring 4a and an inner ring 5. The outer ring 4a has an outer ring raceway 6 on an inner peripheral face, and the inner ring 5 has an inner ring raceway 7 on an outer peripheral face. A plurality of rolling elements 8, 8 are rotatably provided between the outer ring raceway 6 and the inner ring raceway 7 while being held by a retainer 9. In the rolling bearings 3, 3 which are respectively constructed in this manner, the outer ring 4a is fixed to the inside of part of the transmission case, and the inner ring 5 is fixed on the outside of the input rotating shaft 1 or the output rotating shaft 2. Both of these rotating shafts 1 and 2 are rotatably supported within the transmission case by this construction. The outer ring 4a, the inner ring 5, and the rolling elements 8, 8 manufactured of general class 2 bearing steel (SUJ2), have conventionally been used for each rolling bearing 3.

[0004]

Of the rotating shafts 1 and 2, the input rotating shaft 1 is driven to rotate by a drive source 10 such as an engine via a start clutch 11 such as a torque converter or electromagnetic clutch. Furthermore, a drive pulley 12 is provided at a part positioned between the pair of rolling bearings 3, 3 in the middle portion of the input rotating shaft 1, so that this drive pulley 12 and the input rotating shaft 1 rotate synchronously. By displacing one drive pulley plate 13a (on the left in FIG. 2) in the axial direction with a drive actuator 14, the space between the pair of drive pulley plates 13a and 13b constituting the drive pulley 12 can be freely adjusted. That is to say, the width of the groove in the drive pulley 12 can be freely increased or decreased with the drive actuator 14.

[0005]

On the other hand, a driven pulley 15 is provided at a part positioned between the pair of rolling bearings 3, 3 in the middle portion of the output rotating shaft 2, so that this driven pulley 15 and the output rotating shaft 2 rotate synchronously. By displacing one driven pulley plate 16a (on

the right in FIG. 2) in the axial direction with a driven actuator 17, the space between the pair of driven pulley plates 16a and 16b constituting the driven pulley 15 can be freely adjusted. That is to say, the width of the groove in the driven pulley 15 can be freely increased or decreased with the driven actuator 17. An endless belt 18 is spanned between the driven pulley 15 and the drive pulley 12. For this endless belt 18, a metal belt is used.

[0006]

In the belt-type continuously variable transmissions constructed as described above, the drive transmitted from the drive source 10 to the input rotating shaft 1 via the start clutch 11 is transmitted from the drive pulley 12 to the driven pulley 15 via the endless belt 18. Heretofore as the endless belt 18, there is known one wherein drive is transmitted in the push direction, and one wherein drive is transmitted in the pull direction. In either case, the drive transmitted to the driven pulley 15 is transmitted to a drive wheel 21, 21 from the output rotating shaft 2 via a reduction gear train 19, and a differential gear 20. When changing the gear ratio between the input rotating shaft 1 and the output rotating shaft 2, the groove width of the pulleys 12 and 15 is increased or decreased while changing the relationship between the two.

[0007]

For example, to increase the speed reduction ratio between the input rotating shaft 1 and the output rotating shaft 2, the width of the groove in the drive pulley 12 is increased, and the width of the groove in the driven pulley 15 is decreased. As a result, the diameter of the parts of the pulleys 12 and 15 spanned by part of the endless belt 18 is decreased on the pulley 12, and increased on the pulley 15, and speed reduction is performed between the input rotating shaft 1 and the output rotating shaft 2. Conversely, to increase the speed increasing ratio between the input rotating shaft 1 and the output rotating shaft 2 (decrease the speed reduction ratio), the width of the groove of the drive pulley 12 is decreased, and the width of the groove of the driven pulley 15 is increased. As a result, the diameter of the

parts of the pulleys 12 and 15 spanned by part of the endless belt 18 is increased on the pulley 12, and decreased on the pulley 15, and speed increase is performed between the input rotating shaft 1 and the output rotating shaft 2.

[0008]

When operating the belt-type continuously variable transmission constructed and operating as described above, lubricating oil is supplied to each moving part to lubricate each moving part. The lubricating oil employed in the belt-type continuously variable transmission is CVT fluid (including ATF compatible oil). The reason for this is to increase and stabilize the coefficient of friction of the frictional engagement parts of the metal endless belt 18 and the drive and driven pulleys 12 and 15. The CVT fluid is circulated to the frictional engagement parts at a flow rate of at least 300cc per minute to lubricate these frictional engagement parts. Moreover, part of the CVT fluid is passed through the interior of each of the rolling bearings 3, 3 (for example, at a flow rate of at least 20cc per minute) to lubricate the rolling contact parts of the rolling bearings 3, 3. Therefore there is a high possibility that foreign matter such as wear particles generated by wear accompanying frictional between the endless belt 18 and the pulleys 12 and 15, and gear dust generated by friction in the reduction gear train 19, will become mixed with the CVT fluid and enter the interior of these rolling bearings 3, 3. Such foreign matter may damage the rolling contact parts of the rolling bearings 3, 3, and reduce their durability.

[0009]

Therefore, heretofore the bearing size of the rolling bearings 3, 3 has been increased, or the diameter Da of the rolling elements 8, 8 has been increased, to increase the basic dynamic load rating of the rolling bearings 3, 3, and to provide a margin for the life of the rolling bearings 3, 3. However, when in this manner the diameter Da of the rolling elements 8, 8 is increased to maintain the basic dynamic load rating, a thickness T of the outer ring 4a must be reduced (made thinner) to reduce the size and weight of the belt-type continuously variable transmission. Furthermore,

when the rigidity of the transmission case securing the outer ring 4a is low, if the thickness T of this outer ring 4a is reduced in this manner, elastic deformation of the outer ring 4a occurs readily, and an excessive bending stress is applied to the outer ring 4a accompanying the deformation, so that there is a possibility that the life of the rolling bearings 3, 3 will be reduced.

[0010]

For example, in the Proceedings of the Tribology Conference of the Japanese Society of Tribologists (Morioka 1992-10) E-33, pp793-796, it is disclosed that the life of this rolling bearing was reduced by 1/4 to 1/5 when the rolling bearing was operated with a bending stress of 70MPa applied to the raceway ring, in comparison to the case where a bending stress was not applied. Furthermore, it is disclosed that, in order to prevent such a reduction in life, manufacture of the raceway ring from a material wherein a retained compression stress has been applied is effective. However, in order to employ the material wherein such a retained compression stress has been applied, carburized steel must be employed in the raceway ring, and mechanical processes such as shot-peening and the like must be applied to the raceway face of the raceway ring, with the possibility of increased cost.

[0011]

[Problems to be Solved by the Invention]

Recently, in order to ensure the efficiency of belt-type continuously variable transmissions, to suppress noise generated during operation, and to suppress wear of the drive and driven pulleys 12 and 15, and the endless belt 18, the use of a fluid of lower viscosity is under consideration as a CVT fluid. In this case, if standard rolling bearings are employed as the rolling bearings 3, 3 to support the input and output rotating shafts 1 and 2, the possibility of premature flaking is considered to increase. That is to say, the action of vibration in the radial and axial directions accompanying belt fluctuations exacerbates elastic deformation of the outer ring 4a and the inner ring 5, and an excessive bending stress is applied to the outer ring 4a and the inner

ring 5. Accompanying this deformation and excessive bending stress, metal-to-metal contact based on sliding, occurs more readily in the rolling contact parts between the outer ring raceway 6 and the inner ring raceway 7, and the rolling contact surfaces of the rolling elements 8, 8, and the possibility of premature flaking of the outer ring raceway 6, the inner ring raceway 7, and the rolling contact surfaces of the rolling elements 8, 8 increases due to such metal-to-metal contact.

[0012]

That is to say, there are cases where the temperature of the rolling bearing 3 during operation of the belt-type continuously variable transmission may exceed 100°C. At this time the kinetic viscosity of the CVT fluid which enters the interior of the rolling bearing 3 and lubricates the rolling contact parts of the rolling bearing 3 is considerably low at $10mm^2$ per second or less. Moreover, there is also a possibility of a tendency for the amount of CVT fluid supplied to the rolling contact parts to become insufficient. Furthermore, when the rigidity of the transmission case, being the fixed part, is low, the outer ring 4a fixed to the transmission case is readily elastically deformed, and sliding based on differential movement, revolution, and spinning of the rolling elements 8, 8 occurs readily in the rolling contact parts accompanying this deformation. As a result, together with the lack of CVT fluid as described above, the oil film on the rolling contact parts readily breaks up. When the oil film breaks up in such a manner, the outer ring raceway 6 and the rolling contact surfaces of the rolling elements 8, 8 enter an activated state wherein surface fatigue associated with, for example, hydrogen embrittlement flaking due to hydrogen penetration, and metal-to-metal contact, is accelerated, and the possibility of premature flaking increases.

[0013]

On the other hand, according to Hertz' theory of elastic contact, the maximum shear stress under rolling contact is calculated to occur at a depth from the raceway face of approximately 2% of the diameter of the rolling

element. In this case, the thickness of the raceway ring wherein the maximum shear stress occurs is calculated as being semi-infinite. On the other hand, in the case of a standard JIS name and number's rolling bearing, the thickness of the raceway ring tends to be set approximately ten times the depth from the raceway face to the position where the maximum shear stress occurs, that is to say, approximately 20% of the diameter of the rolling element 8. The reason for this is that, when the raceway ring is fixed to a highly rigid part, if the thickness of this raceway ring is approximately 20% of the diameter of the rolling element, the Hertz' theory of elastic contact wherein the thickness of this raceway ring is considered as semi-infinite is established. Moreover, it is considered that experimentally sufficient durability can be maintained. Therefore, in the case of the rolling bearing 3 incorporated in the belt-type continuously variable transmission, if the rigidity of the transmission case is low, the thickness of the outer ring 4a fixed to this transmission case must be increased (made thicker) to ensure durability of the rolling bearing 3. However, simply increasing the thickness of the outer ring 4a in this manner invites increased weight associated with increased size, and increased rolling resistance. Therefore it is not desirable.

[0014]

In Japanese Unexamined Patent Publication No. H10-37951, there is disclosed an invention for improving the permissible high-speed performance of rolling bearings used for machine tools, by increasing the thickness of the outer ring in comparison to the thickness of the inner ring. That is to say, a construction is disclosed wherein ceramic rolling elements are used to thereby reduce the centrifugal force applied to the outer ring, being the fixed raceway ring. Moreover, in order to reduce the centrifugal force generated in the inner ring, being the rotating raceway ring, the thickness of this inner ring is made 2.5mm to 4.0mm, and the thickness of the outer ring is 2.0 to 2.75 times the thickness of the inner ring. However, with this structure, the purpose of making the thickness of the outer ring greater than the thickness of the inner ring is simply to reduce the

centrifugal force by reducing the thickness of the inner ring, and not to prevent elastic deformation of the outer ring fixed to the low-rigidity part. Moreover, since the rolling elements are made of ceramic, increased materials costs and machining costs cannot be avoided. Furthermore, since the thickness of the outer ring is excessive, the rolling contact surfaces of the rolling elements are readily damaged, as described later.

Taking into consideration the above situation, the present invention has been made in order to provide a rolling bearing for use in belt-type continuously variable transmissions wherein damage such as premature flaking and the like to the outer ring raceway 6, the inner ring raceway 7, and the rolling contact surfaces of the rolling elements 8, 8 constituting the rolling contact parts does not occur readily, even when a low-viscosity CVT fluid is employed and the outer ring is fixed to a transmission case of low-rigidity material such as aluminum alloy.

[0015]

[Means to solve the Problems]

The rolling bearing for use in belt-type continuously variable transmissions of the present invention comprises an outer ring, an inner ring, and a plurality of rolling elements.

Of these, the outer ring has an outer ring raceway on its inner peripheral surface.

Moreover, the inner ring has an inner ring raceway on its outer peripheral surface.

Furthermore, the rolling elements are rotatably provided between the outer ring raceway and the inner ring raceway.

The outer ring is fitted into and supported inside a fixed part of a transmission case, and the inner ring is fitted onto and supported on a part which rotates together with a pulley constituting the belt-type continuously variable transmission, such as the end or an intermediate part of input and output rotating shafts, so that the pulley is rotatably supported on the fixed part.

[0016]

In particular, in the rolling bearing for use in belt-type continuously variable transmissions of the present invention, when the minimum thickness (thickness in the radial direction) of the part where the outer ring raceway is provided on the central portion in the axial direction of the outer ring is h, and the diameter of the rolling elements is Da, the relationship $0.4Da \leqq h \leqq 0.8Da$, or more desirably, $0.4Da \leqq h \leqq 0.6Da$, is satisfied.

[0017]

[Operations]

In the rolling bearing for use in belt-type continuously variable transmissions of the present invention constructed as described above, sufficient flaking life can be maintained, even if a low-viscosity CVT fluid is used, and the rolling bearing is incorporated into a low-rigidity transmission case.

That is to say, even when the outer ring is fixed to a low-rigidity transmission case made of aluminum alloy for example, elastic deformation of the outer ring, and application of excessive stress on the outer ring accompanying this deformation can be prevented, without needlessly increasing the thickness of the outer ring. Therefore even in cases where, due to using a low-viscosity CVT fluid, or not circulating a large volume of lubricating oil (for example, a volume greatly exceeding 20cc per minute) inside of the rolling bearing, it is difficult to maintain the strength of an oil film on the rolling contact parts between the outer ring raceway and the inner ring raceway, and the rolling contact surface of the rolling elements, it is possible to prevent metal-to-metal contact in these rolling contact parts, and to maintain sufficient flaking life. It is therefore no longer necessary to increase the size of the rolling bearing in order to maintain the necessary durability, so that the rotating support parts of the input rotating shaft and the output rotating shaft can be made small and light-weight, and turning resistance can be reduced. As a result, the belt-type continuously variable transmission can be reduced in size and weight, and transmission efficiency can be improved.

[0018]

[Embodiments of the Invention]

FIG. 1 shows an example of an embodiment of the present invention. A characteristic of the present invention is that a structure is devised for a rolling bearing 3a for supporting input and output side rotating shafts 1 and 2 (see FIG. 2) for a belt-type continuously variable transmission, so that even when the rigidity of a transmission case is low, sufficient durability of the rolling bearing 3a is maintained. Since the structure and operation of other parts, including the structure shown in FIG. 3, are similar to the heretofore known rolling bearings for belt-type continuously variable transmissions, equivalent parts are denoted by the same reference symbols, and duplicated description is omitted or simplified. Hereunder, the description focuses on characteristic parts of the present invention.

[0019]

In the present example, when the minimum thickness (thickness in the radial direction) of the part where the outer ring raceway 6 is provided on the middle portion in the axial direction of the outer ring 4a is h, and the diameter of the rolling elements 8, 8 is Da, the dimensions of the outer ring 4a are controlled so that the relationship $0.4Da \leqq h \leqq 0.8Da$, or more desirably, $0.4Da \leqq h \leqq 0.6Da$, is satisfied. Furthermore, the width W in the axial direction of the outer ring 4a and the inner ring 5 is controlled within a range which satisfies $1.2Da \leqq Wa \leqq 2.5Da$. In the case of the rolling bearing 3a in the present example, sufficient flaking life can be maintained, even if a low-viscosity CVT fluid is used and the rolling bearing is incorporated into a low-rigidity transmission case.

[0020]

That is to say, even when the outer ring 4a is fixed to a light-weight and low-rigidity transmission case such as of aluminum alloy, elastic deformation of the outer ring 4a, and application of excessive stress on the outer ring 4a accompanying this deformation, can be prevented without needlessly increasing the thickness h of the outer ring 4a. Therefore even in cases where, due to using a low-viscosity CVT fluid, or not circulating a large volume of lubricating

oil (for example, volumes greatly exceeding 20cc per minute) inside of the rolling bearing 3a, it is difficult to maintain the strength of an oil film on the rolling contact parts between the outer ring raceway 6 and the inner ring raceway 7, and the rolling contact surface of the rolling elements 8, 8, it is possible to prevent metal-to-metal contact in these rolling contact parts, and to maintain sufficient flaking life. It is therefore no longer necessary to increase the size of the rolling bearing 3a in order to maintain the necessary durability, so that the rotating support parts of the input rotating shaft 1 and the output rotating shaft 2 can be made small and light-weight, and turning resistance can be reduced. As a result, the belt-type continuously variable transmission can be reduced in size and weight, and transmission efficiency can be improved.

[0021]

If the minimum thickness h of the outer ring 4a exceeds 0.8Da, the rolling elements 8, 8 are not readily assembled into the rolling bearing 3a. That is to say, when the rolling bearing 3a is assembled using automatic assembly equipment, the rolling elements 8, 8 which are normally assembled last, are assembled into the rolling bearing with the outer ring 4a elastically deformed. Therefore, if the minimum thickness h exceeds 0.8Da, the load required to elastically deform the outer ring 4a increases, so that the outer ring 4a and the rolling elements 8, 8 may be readily damaged, and it may no longer be possible to assemble using automatic assembly equipment. On the other hand, if the minimum thickness h is less than 0.4Da, and if the rigidity of the transmission case which secures the outer ring 4a is low, the outer ring 4a is readily elastically deformed, and premature flaking may occur on the outer ring raceway 6 and the inner ring raceway 7, and the rolling contact surfaces of the rolling elements 8, 8.

[0022]

Moreover, the width W in the axial direction of the outer ring 4a and inner ring 5 is desirably as large as possible to prevent elastic deformation of the outer ring 4a and the inner ring 5. However, if the width W is increased,

the mass of the outer ring 4a and the inner ring 5 is also increased. That is to say, if the width W exceeds 2.5Da, the mass of the outer ring 4a and the inner ring 5 become too large, and the transmission efficiency of the belt-type continuously variable transmission may be reduced. On the other hand, if the width W is less than 1.2Da, the rigidity of the outer ring 4a and the inner ring 5 is reduced, and the outer ring 4a and the inner ring 5 may readily elastically deform. It is therefore desirable that the width W be kept within a range of 1.2Da or greater and 2.5Da or less.

[0023]

Furthermore, in the present example, no sealing members are provided in the openings at both ends of the part where the plurality of rolling elements 8, 8 are provided between the inner peripheral surface of the outer ring 4 and the outer peripheral surface of the inner ring 5. However, when there is a high possibility of entry of a significant amount of foreign matter such as wear particles from the drive and driven pulleys 12 and 15 and the endless belt 18 (see FIG. 2), it is desirable that a sealing member be provided, provided that the dimensions in the axial direction of the rolling bearing permit this. As such a sealing member, in addition to a TM seal, a non-contact type seal made of metal plate, or a nitryl seal or acryl seal or fluorine seal of a contact type or a non-contact type and the like can be selected to use in consideration of the temperature in use.

[0024]

Moreover, the construction and the material of the retainer 9 which rotatably holds the rolling elements 8, 8 is not particularly limited. However when the rotational speed in use is particularly high, the use of a crown type retainer made of synthetic resin is desirable to reduce friction between the retainer and the rolling elements, and to suppress the generation of hard wear particles, thus extending life. On the other hand, in cases where rupture of the retainer may occur due to the action of a large variable load, use of a metal waveform retainer is desirable.

[0025]

Furthermore, in the present example, the outer

ring 4a, the inner ring 5, and the rolling elements 8, 8 constituting the rolling bearing 3a are made of class 2 bearing steel (SUJ2) wherein the amount of retained austenite g_R is 5 to 15 volume %. However, when there is a large amount of foreign matter inside the belt-type continuously variable transmission, mixed with the CVT fluid, and passing through the installation space of the rolling elements 8, 8 of the rolling bearing 3a, it is desirable that the steel constituting the outer ring 4a, the inner ring 5, and the rolling elements 8, 8 be carburized or carbonitrided. If the amount of retained austenite in the surfaces of the outer ring 4a, the inner ring 5, and the rolling elements 8, 8 is 20 to 45 volume %, and the surface hardness is approximately (H_R C 62 to 67) with such treatment, damage to these surfaces by foreign matter can be prevented and the durability of the rolling bearing 3a can be increased. Moreover, when the temperature of the rolling bearing 3a in use reaches 150°C or more, it is desirable that a dimension stabilizing treatment which suppresses retained austenite to approximately 0 to 5% be applied to the outer ring 4a, the inner ring 5, and the rolling elements 8, 8. In this case, it is desirable that a heat-resistant rubber be used as the sealing member.

[0026]

Furthermore, in the present example, the internal clearances of the rolling bearing 3a are normal clearances, and the radius of curvature of the cross-section of the outer ring raceway 6 and the inner ring raceway 7 is 0.52 times the diameter of the rolling elements 8, 8 (0.52Da) in all cases. However if the internal clearances and radius of curvature of the cross-section of the raceways 6 and 7 are appropriately controlled (kept small; for example, the radius of curvature of the inner ring raceway 7 is controlled to at least 0.51Da), [the radius of curvature of the outer ring raceway 6 is controlled to 0.535Da,] a backlash in the radial direction and backlash in the axial direction are suppressed, and the contact pressure between the rolling contact surfaces of the rolling elements 8, 8 and the outer ring raceway 6 and inner ring raceway 7 is made uniform, then durability-related performance may be further increased. Moreover, regarding the

rolling bearing 3a, the operation and effects obtained are not limited to the single-row deep-groove type ball bearing as shown in the drawings, and may also be obtained with other types of ball bearings such as angular types, and additionally with cylindrical roller bearings and tapered roller bearings, needle bearings, and other bearings.

[0027]

[Examples]

Next is a description of an experiment conducted to verify the effects of the present invention. In the experiment, as shown in the following Table 1, durability was respectively measured on a total of 14 samples: ten samples (examples 1 through 10) being within the technical scope of the present invention wherein the minimum thickness h of the outer ring 4a was between 0.4 and 0.8 times the diameter Da of the rolling elements (balls) 8, 8 and four samples (comparative examples 1 through 4) being outside the technical scope of the present invention. These samples were based on JIS name-number 6209 (inner diameter d = 45mm, outer diameter D = 85mm, width W = 19mm, ball diameter Da = 11.906mm) and JIS name-number 6310 (inner diameter d = 50mm, outer diameter D = 110mm, width W = 27mm, ball diameter Da = 11.906mm) ball bearings, and were adjusted to the dimensions noted in Table 1 below by respectively varying the outside diameter of each outer ring and the diameter of the balls.

[0028]

[Table 1]

		Outer ring outer diameter D [mm]	Outer ring thickness h [mm]	Ball diameter Da [mm]	h	L_{10} life [hr]	Presence of damage	Other
Example	1	87	5	11.906	0.42Da	500	5/5 normal	6209 base
	2	89	6		0.50Da	500	5/5 normal	
	3	91	7		0.59Da	500	5/5 normal	
	4	93	8		0.67Da	500	1/5 ball damage	
	5	95	9		0.76Da	500	2/5 ball damage	
	6	117	8	19.050	0.42Da	500	5/5 normal	6310 base
	7	119	9		0.47Da	500	5/5 normal	
	8	121	10		0.53Da	500	5/5 normal	

	9	123	11		0.58Da	500	5/5 normal	
	10	125	12		0.63Da	500	1/5 ball damage	
Comp. example	1	85	4	11.906	0.34Da	97	5/5 outer ring flaking	6209 (std)
	2	97	10		0.84Da	125	5/5 ball flaking	6209 base
	3	110	4.5	19.050	0.23Da	84	5/5 outer ring flaking	6310 (std)
	4	113	6		0.31Da	91	5/5 outer ring flaking	6310 base

[0029]

Rolling bearings 3, 3a of the dimensions noted in the Table 1 were respectively incorporated in a belt-type continuously variable transmission as shown in FIG. 2, and employed to rotatably support the input rotating shaft 1 in relation to the transmission case. The arithmetic average roughness Ra of each surface constituting the rolling contact part was between 0.01 and 0.03mm as with normal rolling bearings. Furthermore, the bearing material was standard class 2 bearing steel (SUJ2, hardness = H_R C 60 to 65). Moreover, steel waveform pressing retainers were employed as the retainer 9. Furthermore, the rolling elements (balls) 8, 8 were of SUJ2, through hardening and annealed, and then finished by grinding.

[0030]

A durability test was conducted with a target time of 500 hours under the conditions described below. Following completion of the test, the rolling bearings 3, 3a were dismantled and the component parts of the rolling bearings 3, 3a were checked for damage, and the L_{10} life (rated fatigue life) was calculated. In the current experiment, in order to obtain the durability of the rolling bearings 3, 3 (3a, 3a) incorporated in the rotating support part of the input rotating shaft 1, a sufficient amount (200cc per minute) of lubricating oil (CVT fluid) was supplied to the rolling bearings 3, 3 incorporated in the rotating support part of the output rotating shaft 2. Moreover care was taken to ensure

that rolling bearings 3, 3 not tested would not be damaged before tested rolling bearings 3, 3 (3a, 3a) would be damaged.

Test conditions were as follows..

[0031]

Test equipment: Belt-type continuously variable transmission shown in FIG. 1.

Number of test samples: Five of each sample.

Method of evaluation: Dismantled after 500 hours running, however test discontinued immediately and bearing dismantled if vibration value increased rapidly during the test.

Input torque from engine to input rotating shaft 1: 250N×m (bearings in accordance with or based on JIS name-number 6209), and 500N×m (bearings in accordance with or based on JIS name-number 6310)

Rotational speed of input rotating shaft 1: 6000min-1

Lubricating oil: CVT fluid {dynamic viscosity at 40°C = 35mm2 per sec = 35 x 10-6m2 per sec (35 cSt), viscosity at 100°C = 7mm2 per sec = 7 x 10-6m2 per sec (7 cSt)}

Lubricating oil flow rate: 10cc per minute

Bearing temperature: 120°C

Incidentally, the ratio of engine torque and basic dynamic load rating of rolling bearing was approximately the same for all rolling bearings.

[0032]

The following points were understood from the results of the experiment conducted under the above conditions.

Firstly, in all examples 1 through 10 within the technical scope of the present invention, operation was able to be continued without damage to the rolling bearings 3a until they reached the target 500 hours. Moreover, of these, examination of the raceway surfaces of examples 1 through 3 and examples 6 through 9 after the test revealed that polishing traces remained and the state of lubrication was satisfactory. Furthermore, no damage based on creep was apparent on the outer peripheral surface of the outer ring 4a.

[0033]

On the other hand, damage to the rolling elements (balls) 8, 8 was found in examples 4, 5, and 10. This damage was considered to be due to the fact that the outer ring 4a

did not readily elastically deform when the rolling elements (balls) 8, 8 were assembled into the rolling bearings 3a, due to the thickness h of the outer ring 4a being large enough. Consequently it was understood that the rolling elements 8, 8 should not be readily damaged when assembling such rolling bearings 3a, and it is more desirable that the thickness h of the outer ring be 0.4Da to 0.6Da ($0.4Da \leqq h \leqq 0.6Da$).

[0034]

Moreover, premature flaking occurred (after 84 to 125 hours) in the rolling contact parts, and severe vibration was generated in all the comparative examples 1 through 4 outside the technical scope of the present invention. Furthermore, of these, examination of the raceway surfaces of comparative examples 1 and 3 revealed that part of the polishing traces had not remained, and it was considered that localized metal-to-metal contact had occurred. Moreover, damage to the outer peripheral surface of the outer ring 4 based on creep was apparent, and it was considered that slippage of the rolling elements 8, 8 occurred in the load zone of the outer ring based on this creep. Furthermore, since the thickness h of the outer ring 4 of the comparative example 2 was too great at 0.84Da, it was considered that premature flaking occurred at the point of damage occurring when the rolling elements (balls) 8, 8 were assembled into the rolling bearing 3. Moreover, since the thickness h of the outer ring 4 of the comparative example 4 was too small at 0.31Da, similarly to the comparative examples 1 and 3, damage based on metal-to-metal contact in the rolling contact parts accompanying elastic deformation of the outer ring 4a was apparent on the outer ring raceway 6.

[0035]

[Effects of the Invention]

Since the rolling bearing of the present invention for use in belt-type continuously variable transmissions is constructed and operates as described above, sufficient durability can be maintained even if a low-viscosity CVT fluid is employed, and the outer ring is fixed in a transmission case having a low-rigidity. It is therefore possible to increase the efficiency of belt-type continuously variable

transmissions while maintaining durability.

[Brief Description of the Drawings]

[FIG. 1] A section view similar to FIG. 3, showing an example of an embodiment of the present invention.

[FIG. 2] A simplified section view of a drive system of an automobile incorporating a belt-type continuously variable transmission comprising rolling bearings being the object of the present invention.

[FIG. 3] An enlarged view showing a removed rolling bearing.

[Explanation of Symbols]

1	input rotating shaft
2	output rotating shaft
3, 3a	rolling bearing
4, 4a	outer ring
5	inner ring
6	outer ring raceway
7	inner ring raceway
8	rolling element
9	retainer
10	drive source
11	start clutch
12	driven pulley
13a, 13b	drive pulley plate
14	drive actuator
15	driven pulley
16a, 16b	driven pulley plate
17	driven pulley plate
18	endless belt
19	reduction gear train
20	differential gear
21	drive wheel

[Name of Document] Abstract

[Abstract]

[Problem] Even if a low-viscosity material is used as CVT fluid, and an outer ring 4a is incorporated into a low-rigidity transmission case, durability of a rolling bearing 3a is ensured.

[SOLUTION] When a minimum thickness of a part where the outer ring raceway 4a is provided on a middle portion in the axial direction of an outer ring 6 is h, and a diameter of each rolling element 8, 8 is Da, the relationship $0.4Da \leqq h$ $0.8Da$ is satisfied. As a result, even when the outer ring 4a is fixed in a transmission case of low-rigidity, elastic deformation of the outer ring 4a can be prevented without needlessly increasing the thickness h of the outer ring. Also, application of excessive stress on the outer ring accompanying this deformation can be prevented to solve the above problem.

[Selected Figure] Fig. 1.

FIG. 1

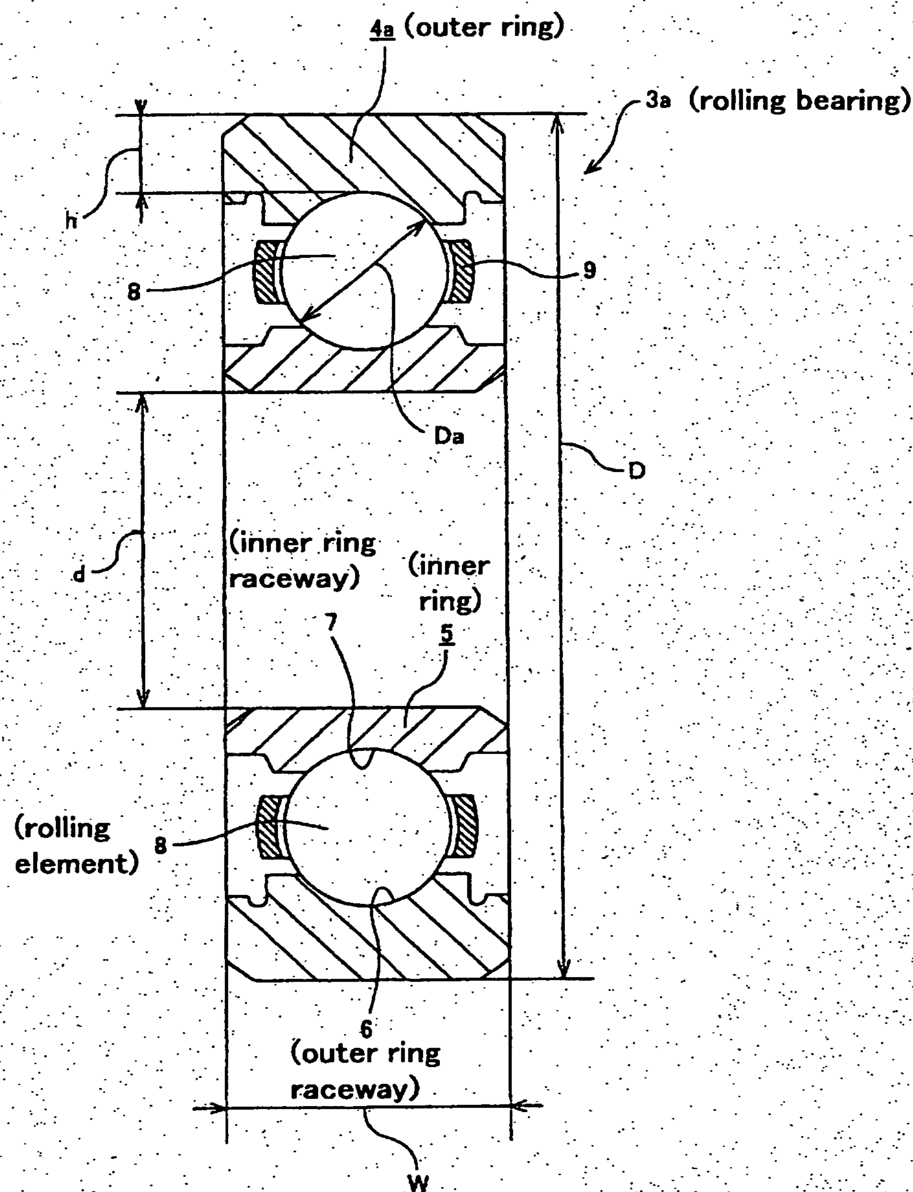
4a (outer ring)
3a (rolling bearing)
h
8
9
Da
D
d
(inner ring raceway)
(inner ring)
5
7
(rolling element)
8
6
(outer ring raceway)
W

FIG. 2

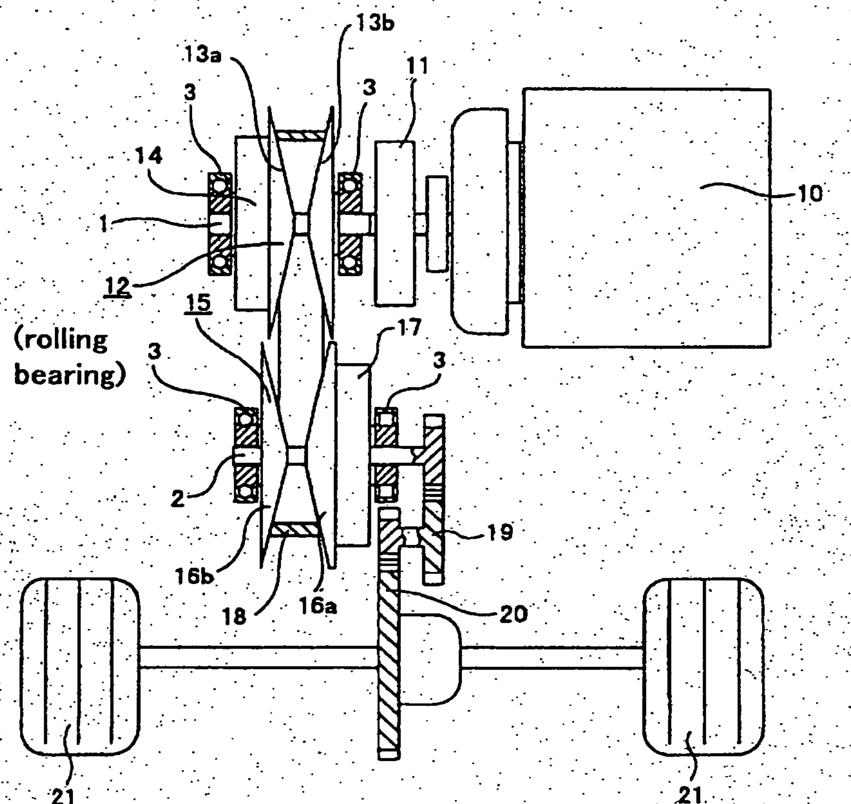
13b
13a
3
11
3
14
10
1
12
15
17
(rolling
bearing)
3
3
2
19
16b
18
16a
20
21
21

FIG. 3